TITLE: A New On-Board Gauge Calibration Process for Aircraft Engine Testing

A New On-Board Gauge Calibration Process for Aircraft Engine Testing

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1 ABSTRACT

A new procedure for calibrating on-board gauges on the blade assemblies of turbomachines is proposed in this paper. The calibration of an engine gauge, determined *a priori* during a component dynamic test, makes it possible to establish the levels of dynamic stress on the entire instrumented airfoil (based on the readings output by the gauge during the engine test). The method proposed is based on the use of holographic dynamic measurements enabling the qualification of a finite element model in relation to the component test. The calibration relations are then determined based on the qualified model. The procedure proposed enables an increase in the reliability of the calibration relations by making it possible to take into account in the model the real conditions of the engine test (thermal, centrifugal and aerodynamic loads, and the shapes of dynamic modes of the complete blade assembly). Similarly, it enables a drastic reduction in the costs relating to the component tests required to establish these relations.

2 INTRODUCTION

The procedures for designing blade assemblies for turbo engines use different types of analysis, in particular to guarantee the safety of the engine. With regard to dynamic considerations, these analyses are based both on the results of tests and on the results of calculations. Whatever the design approach, the certification authorities require engine tests in order to qualify the machine before use.

Qualification tests consist in measuring the levels of dynamic stress during operation of every flight envelope, in particular for all rotating parts. To do so, strain gauges are installed on the stators or rotor wheels of the qualification engine. These on-board gauges are called engine gauges. Based on the measurements made during operation, the dynamic stress status is determined for the entire blade assembly based on **calibration factors,** i.e. the **relations governing the transfer of strain or stress between engine gauges and the gauges distributed over the airfoil**. These relations are established during component dynamic tests carried out beforehand. After the qualification test (during operation), the maximum stress on the airfoil (deduced from the engine gauges using the calibration factors) is then compared with the admissible dynamic stress determined by numerical means: the real dynamic margin is then estimated.

This procedure for determining the real dynamic margin requires data on:

- the static stress field obtained by finite element calculation,
- the distribution of dynamic stress per mode, obtained by component testing or by finite element calculation,
- the measurements concerning dynamic engine stress during operation output by the engine gauge.

The engine qualification test is carried out at the end of the design cycle. At this point in time, if the dynamic margins are insufficient, re-design is required as well as a new qualification test, thereby increasing the cost of design of the engine.

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The quality with which calibration factors are evaluated is thus vital. This implies expensive, in-depth component dynamic testing. In this paper we propose to reduce the time and the costs involved in determining these calibration factors, and to increase the precision and the frequency band of validity, by using dynamic field measurements of holographic type **[SMI.99]** and results obtained by finite element calculation.

3 DESIGN CRITERIA OF BLADE ASSEMBLY FOR HIGH CYCLE FATIGUE : BASIC PRINCIPLES

Here we briefly review a few basic concepts to make it easier to understand our paper.

Consider a blade of a rotor wheel in a fluid flow, with steady and unsteady components (whose angular frequency is similar to that of the fundamental mode of vibration of the blade), rotating at a given speed.
The steady component is due to the difference in the downstream and upstream pressures of the wheel in question. The unsteady component is mainly related to the wake of the stator wheel located upstream of the rotor wheel in question, as well as to the other obstacles in the aerodynamic flow, upstream and downstream of the wheel in question.
For a given temperature, the blade is therefore subject to a static load caused, on the one hand, by the centrifugal force resulting from the rotation of the wheel around the axis of the engine, and on the other hand, by the steady component of the aerodynamic pressure. The unsteady component of the aerodynamic pressure generates a purely dynamic load on the airfoil (see Figure 1).

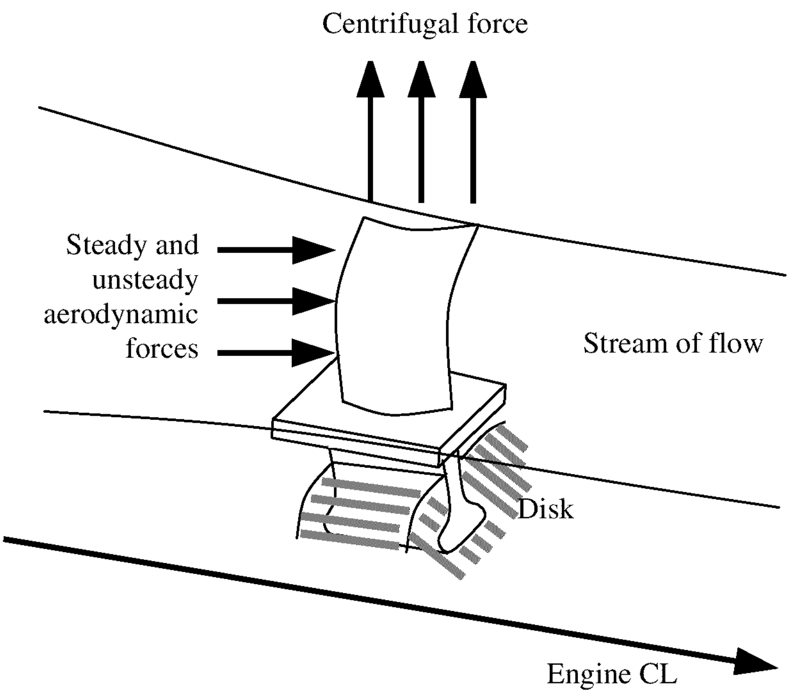

Figure 1: Loads applied to a blade rotating in a fluid flow

The blade is designed for high cycle fatigue such that it has an infinitely long service life. This means that the level of stress obtained on any point of the blade under the combined static and dynamic load does not exceed the fatigue limit criteria **[LEM.96]**. The fatigue limit is experimentally obtained by dynamic tests on a test piece with a static preload. The fatigue limit stress pairs are sought for various static/dynamic load ratios (Woehler curves). Based on these limit values, a Goodman diagram is established for a given number of cycles (a cycle corresponding to a period of the dynamic load) and defines the area of the admissible static and dynamic stress pairs corresponding to an infinite service life (Figure 2).

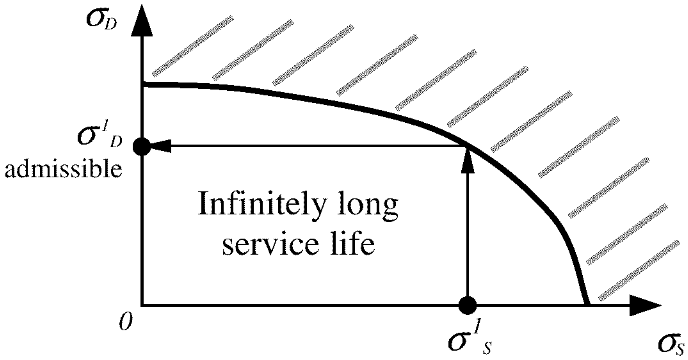

Figure 2: Goodman diagram

The Goodman curve makes it possible to relate an admissible dynamic stress σ^1_D to a given static stress σ^1_S. In the dimensioning phase, the static stress factors are correctly evaluated by finite element calculation, since the related load is correctly characterised. On the other hand, characterising the dynamic stress generated by the unsteady aerodynamic pressure is a more delicate matter. This is because parameters are involved that are difficult to evaluate, such as the characterisation of the unsteady pressure fields caused by the different obstacles upstream and downstream of the wheel in question, as well as their periodicity, mechanical and aerodynamic damping, and the shape of the high-level modes of vibration. This is why these dynamic stress factors must be measured during the qualification test. These measurements are the subject of a special procedure described in the following paragraph.

4 CURRENT CALIBRATION PROCEDURE FOR ENGINE GAUGES

Consider a blade of a given stator or rotor wheel. In the case of blades formed by sectors for certain stator wheels and of blisks, the calibration principle is similar. The calibration phase for engine gauges is carried out on the basis of component dynamic tests. This operation breaks down into two types of tests: stress distribution measurement and calibration tests of the engine gauges.

In the 2 types of tests, the airfoil to be characterised (rotor or stator wheel) is fastened into position with a test jaw (which in turn is fastened onto a test bench). The jaw is designed in order to reproduce as faithfully as possible the boundary conditions during operation. The blade is instrumented with strain gauges (see Figure 3). The excitation applied is of scanned sine-wave type, obtained by means of an electrodynamic vibrating table, an electrodynamic shaker, a piezoelectric shaker, or a siren type shaker (a jet of pulsed air). For each resonance frequency, the signals output by the strain gauges are processed using a spectrum analyser.

4.1 Stress distribution measurement

This measurement is carried out on a single reference blade, for which the measurements are designed to precisely define the stress distribution on the blade for each excited mode of vibration. Although only carried out once, this test is extremely expensive because of the high density of gauges used. The instrumentation of the airfoil can comprise more than a hundred strain gauges: some are located in the middle of the airfoil (low-frequency modes) while most are located along the edges where strains mainly appear on high-frequency modes (see Figure 3). This measurement makes it possible to determine for each mode a strain transfer relation between the instrumented points (and directions). This relation is defined based on the strains $d_{j\nu}$

observed at point j on mode v. These values are combined in the strain transfer matrix D defined by expression (1).

$$D = \begin{bmatrix} d_{11} & d_{12} & \cdots & d_{1v} & \cdots \\ d_{21} & d_{22} & \cdots & d_{2v} & \\ \vdots & \vdots & \ddots & \vdots & \\ d_{j1} & d_{j2} & \cdots & d_{jv} & \\ \vdots & & & & \ddots \end{bmatrix} \tag{1}$$

This strain transfer matrix can be normalised in relation to one of the instrumented points and forms the transfer matrix $\overline{D}^j$, normalised in relation to point j, defined by expression (2).

$$\overline{D}^j = \left[\overline{D}_1^j \quad \overline{D}_2^j \quad \cdots \quad \overline{D}_v^j \quad \cdots\right] = \begin{bmatrix} d_{11}/d_{j1} & d_{12}/d_{j2} & \cdots & d_{1v}/d_{jv} & \cdots \\ d_{21}/d_{j1} & d_{22}/d_{j2} & \cdots & d_{2v}/d_{jv} & \\ \vdots & \vdots & \ddots & \vdots & \\ 1 & 1 & \cdots & 1 & \\ \vdots & & & & \ddots \end{bmatrix} \tag{2}$$

This transfer relation, characterised by matrix $\overline{D}^j$ enables the expansion of the strain values for each point instrumented during the distribution test, based on the value of strain at point j, at right angles with an engine gauge or a reference gauge for example.

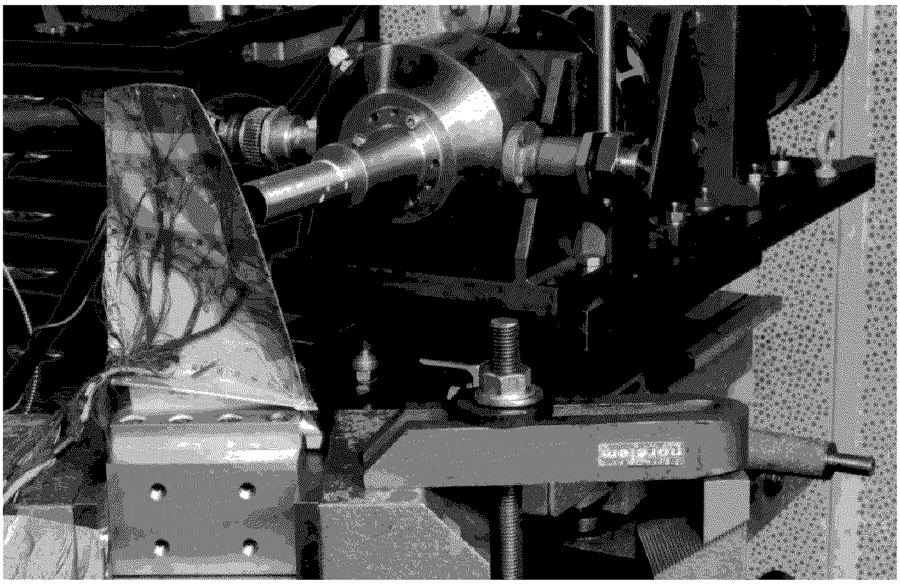

Figure 3: Measurement of stress distribution by strain measurement on a HP compressor blade - excitation by a jet of pulsed air

4.2 Calibration of engine gauges

This test is carried out for each airfoil instrumented with engine gauges. The instrumentation for these component tests can be limited to a few engine gauges and a dozen reference gauges. All these gauges (engine and reference gauges) are placed in positions chosen from among those used when stress distribution is precisely measured. This calibration test is necessary because engine gauges are subject to a special

instrumentation procedure. Engine gauges must be capable of meeting criteria of thermal resistance and static load resistance of the levels caused by the rotation of the wheel. To do so, they are fastened onto the airfoil and protected by a bonding agent. Because of the fastening method and the position of the gauges, this type of instrumentation requires on-site calibration. Calibration is carried out in relation to reference gauges with a known level of sensitivity. The calibration relation is defined between the engine gauge (measurement of strain d^{jm}) and a reference gauge (measurement of strain d^{jr}) using the calibration coefficient α:

$$d^{jr} = \alpha \cdot d^{jm} \tag{3}$$

4.3 Calibration factors

Calibration factors are established, based on distribution factors (defined by relation (2)) and the calibration values for the engine gauges (relation (3)). In this way, for a mode ν of angular frequency ω_ν, the dynamic strain values $d(\omega_\nu)$ are deduced for the airfoil (where are located the gauges instrumented during the distribution test), based on the value measured on the engine gauge $d^{jm}(\omega_\nu)$, by means of relation (4):

$$d(\omega_\nu) = \begin{bmatrix} d_1(\omega_\nu) \\ d_2(\omega_\nu) \\ \vdots \\ d_{jr}(\omega_\nu) \\ \vdots \end{bmatrix} = \overline{D}_\nu^{jr} \cdot d^{jr}(\omega_\nu) = \overline{D}_\nu^{jr} \cdot \alpha_\nu \cdot d^{jm}(\omega_\nu) \tag{4}$$

The calibration procedure is performed for all the blades instrumented with engine gauges and for the blade that was used to measure the distribution of dynamic stress. After this procedure, the blades are installed on the qualification engine. Based on the calibration factors (i.e. the calibration coefficients for the engine gauges as well as the strain transfer relations) and on the static stress calculated by the finite element method, the allowable dynamic stress (scope limit) is determined for each mode of vibration and for each engine gauge. During the engine qualification test, the levels measured by the engine gauges are monitored to guarantee the integrity of the structure. These dynamic levels are then analysed and the safety margins are deduced for all the instrumented wheels. The engine is qualified if these safety margins are large enough. If not, a new design is required for the components in question.

5 PROPOSED CALIBRATION PROCEDURE

The calibration phase comprises two types of component tests: measurement of stress distribution (an expensive test requiring a high level of instrumentation) and calibration tests of the engine gauges. In this paper, we propose to use holographic measurements of field type (Figure 4) in order to qualify the finite element model. This qualification of the FE model is based on a quantitative comparison of the modal displacement fields both measured and calculated. The qualified model is then used to determine the stress distribution in the airfoil.

Holographic measurement techniques are frequently used in industry and are the subject of research work dedicated to improving identification and correlation procedures **[FOL.00] [HUM.99] [LEP.01]**. The technique for acquiring dynamic data by this means of measurement is based on the principle of holographic interferometry. The surface to be measured is illuminated by means of a continuous or pulsed wave laser. The acquisition of optical data is obtained by a CCD video camera (Figure 4). A hologram of the structure is obtained by interference between the reflected laser beam and a reference laser beam. Based on the interference of two holograms of the structure at 2 different instants in time, the requisite displacement data can be obtained.

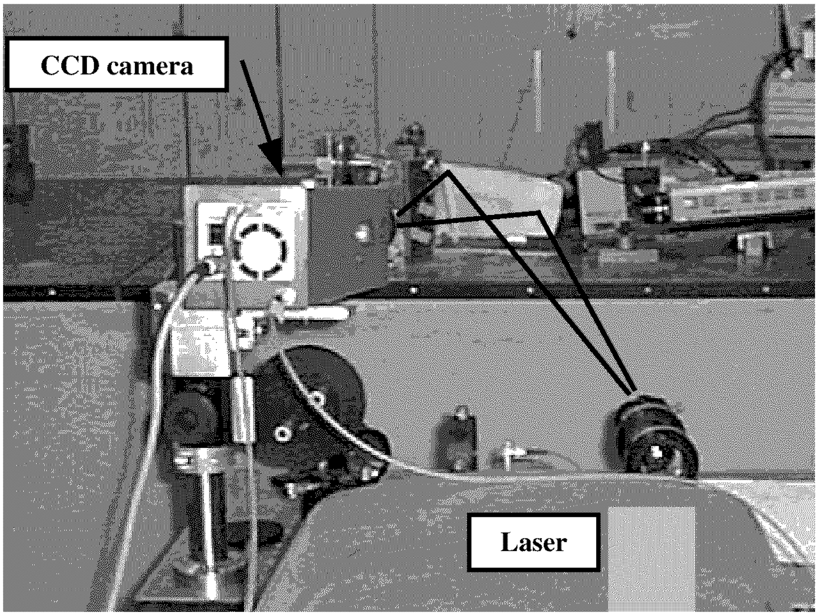

Figure 4: Dynamic measurement by laser holography

This new procedure has a number of advantages:

- The behaviour of the blade is not modified by the instrumentation since measurement is made by optical readings.
- Since stress distribution is obtained by finite element calculation, maximum modal stress can be obtained and the effects of steady centrifugal, thermal and aerodynamic loads can be precisely taken into account in dynamic stress transfer relations, at the operating level of the engine in question, as well as the shape of the normal mode with nodal diameters appropriate to the excitation.
- A component dynamic test of quantitative holography type is relatively inexpensive in relation to a stress distribution test, since the blade is not subject to lengthy, pains-taking instrumentation.

5.1 Mathematical model hypotheses

In general, blade assemblies are structures with little damping (damping is limited to around 1% at most). For this reason, within the framework of a component test under periodic excitation, it is legitimate to consider only the contribution of the excited mode in the dynamic response of the blade near the resonance frequency. Given the hypothesis of a material with linear behaviour and small displacements, the dynamic equilibrium equation for the blade formulated for the sinusoidal excitation of angular frequency ω results in relation (5) :

$$\left[K + j\omega B - \omega^2 M\right] y(\omega) = f(\omega) \tag{5}$$

where K, B, and M are respectively the stiffness, damping and mass matrices for the structure, and $y(\omega)$ and $f(\omega)$ are the vectors of harmonic displacement response and harmonic force applied to the structure.

After conventional breakdown of the harmonic response of the structure on a truncated basis of m fundamental modes of vibration for the conservative system (relation (6)), the dynamic equilibrium of the structure is governed by relation (7):

$$y(\omega) = Y_m \, c(\omega) \tag{6}$$

$$\left[\Lambda_m + j\omega \beta_m - \omega^2 I_m\right] c(\omega) = Y_m^T f(\omega) \tag{7}$$

in which Λ_m, Y_m are respectively the spectral and modal matrices of the m first fundamental modes of vibration of the conservative system, normalised per unit of generalised mass, $c(\omega)$ is the vector of the generalised co-ordinates and β_m the generalised damping matrix.

Based on the hypothesis that the only contribution is that of the excited mode ν (characterised by its angular frequency ω_ν and its eigenvector y_ν), the harmonic response of the structure near resonance angular frequency ω_ν is given by relation (8):

$$y(\omega) = \frac{y_\nu y_\nu^T f(\omega)}{\omega_\nu^2 - \omega^2 + j\omega\beta_\nu} \tag{8}$$

where $\beta_\nu = 2\, a_\nu\, \omega_\nu$ is the generalised damping of mode ν and a_ν the reduced damping coefficient.

At resonance angular frequency ($\omega = \omega_\nu$), the harmonic response during displacement of the structure is defined by relation (9):

$$y(\omega_\nu) = \frac{y_\nu y_\nu^T f(\omega_\nu)}{j\omega_\nu\, \beta_\nu} \tag{9}$$

The dynamic stress field σ is deduced from the dynamic displacement field y of the structure by means of a relation of linear dependence. At angular frequency ω_ν, it is defined by relation (10):

$$\sigma(\omega_\nu) = D\,\varepsilon(\omega_\nu) = D\,B\;y(\omega_\nu) \tag{10}$$

where D is the elasticity matrix of material, B is the derived matrix of the shape functions, and $\varepsilon(\omega_\nu)$ the dynamic strain field at angular frequency ω_ν.

6 RESULTS OF CORRELATION BETWEEN CALCULATION AND TEST

In order to illustrate the interest of the proposed procedure, two results of correlation calculations/tests are presented for an HP compressor blade (not illustrated in this paper). First of all, we compare the modal stress fields calculated and measured by strain measurement. Secondly, we compare the modal displacements calculated by finite elements and measured by holography. It should be noted that the blades are of the same type. The same finite element model is used in both cases for numerical prediction. Only the boundary conditions are different.

6.1 Correlation criteria

Two comparison criteria are used here in order to compare the calculation results with the results of tests at the instrumented points (in the direction of measurement). One criterion is based on the relative error E_f committed when predicting eigenfrequencies, and a second criterion quantifies the colinearity (or similitude of shape) of the vectors measured and calculated (MAC criterion). Matching between the predicted and observed modes is evaluated by this shape criterion.

- Relative error on frequency E_f

$$E_f = \frac{\left| f_{cal} - f_{mes} \right|}{f_{mes}} \tag{11}$$

where f_{mes} (f_{cal}) is the measured (calculated) eigenfrequency.

- Modal assurance criterion (or matching) on displacement MAC_y, on stress MAC_σ :

$$MAC_y = \frac{\left(y_{cal}^T \cdot y_{mes}\right)^2}{\left\|y_{cal}\right\|^2 \left\|y_{mes}\right\|^2}, \quad MAC_\sigma = \frac{\left(\sigma_{cal}^T \cdot \sigma_{mes}\right)^2}{\left\|\sigma_{cal}\right\|^2 \left\|\sigma_{mes}\right\|^2} \tag{12}$$

where y_{mes} (y_{cal}) and σ_{mes} (σ_{cal}) are the measured (calculated) displacement and stress eigenvectors.

Remark: the value of the MAC criterion is 1 for two colinear vectors, 0 for two orthogonal vectors.

6.2 Strain measurement

Dynamic strain distribution is measured using single-direction strain gauges distributed over the entire airfoil. The excitation of the airfoil is of siren type in the low frequency band and of piezo-electric type in the medium frequency band. 25 modes were measured. The amplitude and the phase of strain were read for each gauge and each mode of resonance.

The result of matching the measured modes and calculated modes, obtained by the criterion MAC_σ, is presented in Figure 5 for the 13 first modes. Only 47 gauge values were used in estimating this criterion. The relative error on the corresponding eigenfrequencies is illustrated in Figure 6.

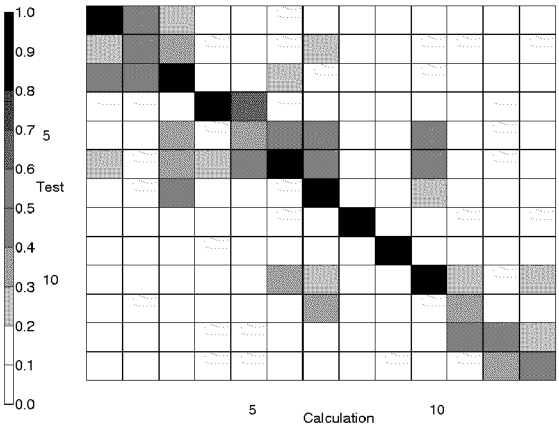

Figure 5: Matching matrix (MAC_σ) between measured modes and calculated modes

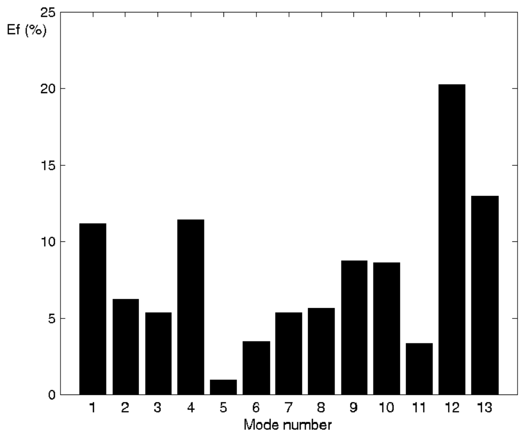

Figure 6: Relative error (%) in frequency (E_f) between matched measured and calculated modes

As can be seen in Figure 5, 8 modes are matched with significant MAC_σ values, higher than 0.7. Mode nos. 2 and 5 (corresponding to the first twists of the blade) were not correctly observed by the measuring gauges selected. The relative errors in eigenfrequency obtained with the matched modes are around 12% at the most.

The stress modeshapes calculated and measured are presented in Figure 7 for mode no.10. For guidance, gauges 1 to 47 describe the contour of the blade.

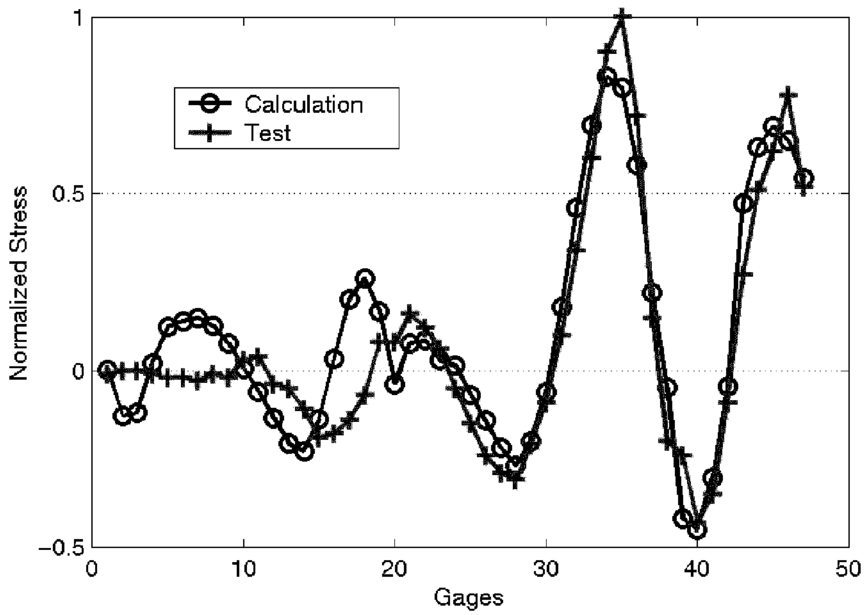

Figure 7: Stress modeshapes calculated and measured for mode no.10

The matching criterion value for mode no.10 is 0.86.

6.3 Holographic measurement

Holographic measurements enable the characterisation of 26 fundamental modes of vibration. Excitation is of piezo-electric type. Holographic images comprise 250,000 measurement points (or pixels). After identification of the geometric transformation between the measurement locator and the calculation locator, the adaptation between measured and calculated data is carried out by projecting the measurement values onto the corresponding partition of the finite element meshing. This operation implies reducing the number of data measured to the number of nodes in the model, i.e. between a hundred and a few thousand nodes. This adaptation operation uses a procedure of geometric smoothing, in which a measured displacement projected onto a node in the mesh is obtained by a combination of the values of the pixels in the vicinity of the node in question.

The entire adaptation procedure for the measurement and calculation data, as well as the calculation of the correlation criteria, are performed using an in-house Snecma Moteurs operational tool.

The result of the matching between measured and calculated modes, obtained using the MAC_y criterion, is presented in Figure 8 for the 26 modes measured. The matching criterion is calculated for all the nodes of the mesh of the airfoil corresponding to the surface measured. The relative error on the corresponding eigenfrequencies is illustrated in Figure 9.

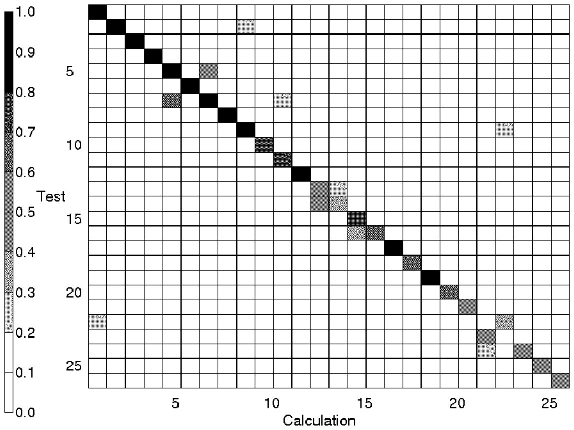

Figure 8: Matching matrix (MAC_y) for measured and calculated modes

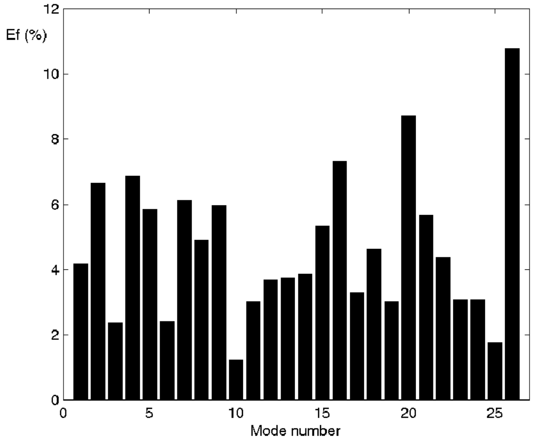

Figure 9: Relative error (%) in frequency (E_f) between matched measured and calculated modes

Examination of the modal assurance criterion indicates that the correlation between the calculated and measured modes is excellent. 17 modes are matched with a MAC_y value higher than 0.7. Given the number of nodes on which the criterion is evaluated, these results reflect the aptitude of the mathematical model to correctly represent the shapes identified by measurement.

Only two measured modes have a MAC_y value of less than 0.5, i.e. measured mode nos.14 and 22. The relative errors in eigenfrequency obtained on the matched modes are around 11% at the most.

For guidance, Figures 10 and 11 show the measured displacement fields on the airfoil (projected onto the nodes of the mesh) and calculated in relation to mode no.20.

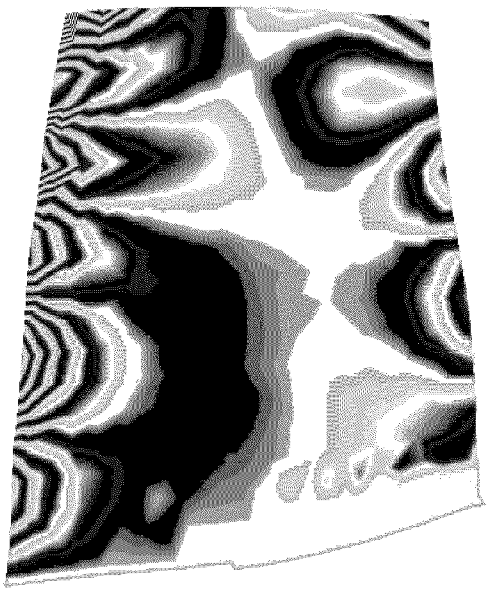

Figure 10: Measured displacement field of mode no.20, projected onto the nodes of the model

Figure 11: Calculated displacement field of mode no.20

The corresponding MAC_y value is 0.83.

7 CONCLUSIONS

Comparison between the model and stress distribution tests based on strain measurement enables the correct correlation of 8 modes of vibration. For uncorrelated measured modes, it is difficult to relate the modes output by calculation with any certainty.
The comparison between the model and holographic measurements makes it possible to correctly correlate all the modes. The wealth of information available via dynamic field measurements (in this case measurement by holographic interferometry) enables non-conflicting matching between model measurement and prediction.
The difficulty in correlating modes 10 to 25 observed in the stress distribution test by strain measurement is due to several factors:

- Modification of the characteristics of the structure: a gauge bonded onto the structure and its connector technology introduce local disturbances of stiffness, mass and damping that are not taken into account in the mathematical model. This structural modification, which is proportional to the density of instrumentation, has no effect on the first global modes but has a considerable impact on the eigen-shape of the high-level modes of vibration.
- Dispersion in gauge sensitivity (dispersion due to instrumentation, and to sensitivity tolerances).
- Errors in adaptation between measured and calculated values: calculated stress values are sensitive to the size of the elements. A variation in the position between a gauge and the element of the corresponding mesh introduces an additional error, in particular in areas of the airfoil with high stress gradients.

Optical measurement by holography enables the characterisation of modal displacement fields without disturbing the structure and provides a dense quantity of data, of much greater wealth than that produced by calculation. Using this information requires an adaptation procedure in order to condense measurements onto the nodes of the model. This special operation has proved to be extremely robust.

The results presented clearly show the advantage in using optical field measurements, which, because of the quality and wealth of the data output, enable clear, non-conflicting matching between measured and calculated modes of vibration.

The use of a qualified model for component tests based on optical measurements makes it possible to take engine operating conditions into account in estimating modal stress factors, i.e. the influence of thermal, centrifugal, and steady aerodynamic loads. Similarly, these stress factors can be calculated in the case of normal modes with nodal diameters (which can be excited by the identified harmonics of the engine speed), which is difficult to reproduce in a component test. Access to the maximum modal stress is another advantage of the calibration method using a model validated by component tests. This is because in the case of strain measurement, the maximum stress measured is determined only at instrumented points.

The procedure proposed to estimate the distribution of dynamic stress is based on the qualification of a finite element model by correlation with holographic interferometry measurements, which increases the reliability of the calibration factors of the engine gauges. In addition, the costs incurred and the time spent in the component tests related to this procedure are considerably reduced compared with a conventional procedure based on strain measurement.

This approach of calibration methodology contributes to improve the quality of prediction of dynamic behaviour, particularly the fatigue life prediction of highly loaded components, and therefore to increase the safety of flights.

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